Notice No. 7

Rules and Regulations for the Classification of Special Service Craft, July 2014

The status of this Rule set is amended as shown and is now to be read in conjunction with this and prior Notices. Any corrigenda included in the Notice are effective immediately.

Issue date: May 2015

Amendments to	Effective date
Part 11, Chapter 2, Scope and Sections 3 & 4	1 July 2015
Part 13, Chapter 1, Sections 2, 3 & 4	1 July 2015

Part 11, Chapter 2

Shafting Systems

Effective date 1 July 2015

■ *Scope*

The requirements of this Chapter relate, in particular, to formulae for determining the diameters of shafting for main propulsion installations, but requirements for couplings, coupling bolts, keys, keyways, sternbushes and other associated components are also included. The diameters may require to be modified as a result of alignment considerations and vibration characteristics, *see* Pt 13, or the inclusion of stress raisers, other than those contained in this Chapter.

Alternative calculation methods for determining the diameters of shafting for main propulsion and their permissible torsional stresses will be considered by LR. Any alternative calculation method is to include all relevant loads on the complete dynamic shafting system under all permissible operating conditions. Consideration is to be given to the dimensions and arrangements of all shaft connections. Moreover, an alternative calculation method is to take into account design criteria for continuous and transient operating loads (dimensioning for fatigue strength) and for peak operating loads (dimensioning for yield strength). The fatigue strength analysis may be carried out separately for different load assumptions, for example as given below.

Shafts complying with the applicable Rules in Pt 11, Ch 2 and Pt 13 satisfy the following:

(a) Low cycle fatigue criterion (typically <10^4), i.e. the primary cycles represented by zero to full load and back to zero, including reversing torque, if applicable. This is addressed by the formulae in Pt 11, Ch 2,4.2, 4.4 and 4.5.

(b) High cycle fatigue criterion (typically >10^7), i.e. torsional vibration stresses permitted for continuous operation as well as reverse bending stresses and the accumulated fatigue due to torsional vibration when passing through a barred speed range or any other transient condition with associated stresses beyond those permitted for continuous operation. This is addressed by the formulae in Pt 13, Ch 1,3.2. The influence of reverse bending stresses is addressed by the safety margins inherent in the formulae from Pt 11, Ch 2,4.2, 4.4 and 4.5.

■ *Section 3*
Materials

3.1 Materials for shafts

3.1.2 The specified minimum tensile strength of forgings for shafts is to be selected within the following general limits:

(a) Carbon and carbon-manganese steel − 400 to 760 N/mm^2. *See also* 4.4.3.

(b) Alloy steel main propulsion shafting:

- (i) not exposed to seawater − not exceeding 800 N/mm^2,
- (ii) for other forgings - not exceeding 1100 N/mm^2.

~~*3.1.2*~~ *3.1.3* Where it is proposed to use alloy steel forgings, particulars of the chemical composition, mechanical properties and heat treatment are to be submitted for approval. ~~For main propulsion shafting, not exposed to sea water, in alloy steels, the specified minimum tensile strength is not to exceed 800 N/mm^2 and for other forgings is not to exceed 1100 N/mm^2.~~

3.1.4 Where shafts may experience vibratory stresses close to the permissible stresses for transient operation, the materials are to have a specified minimum tensile strength of 500 N/mm^2.

3.1.5 Where materials with greater specified or actual tensile strengths than the limitations given above are used, reduced shaft dimensions or higher permissible vibration stresses are not acceptable when derived from the formulae used in sub-Sections 4.2, 4.4, 4.5 and Pt 13, Ch 1,3.2.

Existing paragraphs 3.1.3 *to* 3.1.5 *have been renumbered* 3.1.6 *to* 3.1.8.

■ *Section 4*
Design and construction

4.2 Intermediate shafts

4.2.1 The diameter, *d*, of the intermediate shaft is to be not less than:

$$d = F\,k\,\sqrt[3]{\frac{P}{R}\left(\frac{560}{\sigma_u + 160}\right)} \quad \text{mm}$$

where

k = 1,0 for shafts with integral coupling flanges complying with 4.8 or shrink fit couplings
= 1,10 for shafts with keyways, tapered or cylindrical connections, where the fillet radii in the transverse section of the bottom of the keyway are not less than 0,0125*d*
= 1,10 for shafts with transverse or radial holes (d_h) where the diameter of the hole does not exceed 0,3*d*
= 1,20 for shafts with longitudinal slots ~~having a length of not more than 1,4*d* and a width of not more than 0,2*d* where *d*, is determined with *k* = 1,0~~, *see* 4.2.7

F = 95 for turbine installations, electric propulsion installations and diesel engine installations with slip type couplings
= 100 for other diesel engine installations

P and *R* are as defined in Part 9 (losses in gearboxes and bearings are to be disregarded)

σ_u = specified minimum tensile strength of the shaft material, in N/mm^2.

4.2.5 For shrink fit couplings, *k* refers to the plain shaft section only. Where shafts may experience vibratory stresses close to the permissible stresses for continuous operation, an increase in diameter to the shrink fit diameter is to be provided, e.g. a diameter increase of 1 to 2 per cent and a blending radius as described in 4.8.

4.2.6 Keyways are in general not to be used in installations with a barred speed range.

4.2.7 The application of *k* = 1,20 is limited to shafts with longitudinal slots having a length of less than 0,8d_o and a width greater than 0,15d_o and a diameter of central hole d_i of less than 0,7d_o, *see* 4.5. The end rounding of the slot is not to be less than half the width. An edge rounding should preferably be avoided as this increases the stress concentration slightly. The values of C_K, *see* Table 1.3.1 in Pt 13, Ch 1, are valid for 1, 2 and 3 slots, i.e. with slots at 360, 180 and 120 degrees apart respectively.

Existing paragraph 4.2.5 *has been renumbered* 4.2.8.

4.4 Srewshafts and tube shafts

4.4.3 The diameter, d_p of the protected forged steel screwshaft immediately forward of the forward face of the propeller boss or, if applicable, the forward face of the screwshaft flange, is to be not less than:

$$d_p = 100k\,\sqrt[3]{\frac{P}{R}\left(\frac{560}{\sigma_u + 160}\right)} \quad \text{mm}$$

where

k = 1,22 for a shaft carrying a keyless propeller fitted on a taper, or where the propeller is attached to an integral flange, and where the shaft is fitted with a continuous liner, a coating of an approved type, or is oil lubricated and provided with an approved type of oil sealing gland
= 1,26 for a shaft carrying a keyed propeller and where the shaft is fitted with a continuous liner, a coating of an approved type, or is oil lubricated and provided with an approved type of oil sealing gland

P and *R* are as defined in Part 9 (losses in gearboxes and bearings are to be disregarded)

σ_u = specified minimum tensile strength of the shaft material, in N/mm^2 but is not to be taken as greater than 600 N/mm^2, *see* 3.1.4.

4.4.5 The diameter of the portion of the screwshaft and tube shaft forward of the length required by 4.4.4 to the forward end of the forward stern tube seal is to be determined in accordance with 4.4.3 with a *k* value of 1,15. The change of diameter from that determined with *k* = 1,22 or 1,26 to that determined with *k* = 1,15 should be gradual, *see* 4.8.

4.4.6 Screwshafts which run in sterntubes and tube shafts may have the diameter forward of the forward stern tube seal gradually reduced to the diameter of the intermediate shaft. Abrupt changes in shaft section at the screwshaft/tube shaft to intermediate shaft couplings are to be avoided, *see* 4.8.

4.8 ~~Flange connections of couplings~~ Couplings and transitions of diameters

4.8.7 Transitions of diameters are to be designed with either a smooth taper or a blending radius. In general, a blending radius equal to the change in diameter is recommended.

Part 13, Chapter 1

Torsional Vibration

Effective date 1 July 2015

■ *Section 2* Details to be submitted

2.1 Particulars to be submitted

2.1.3 Enginebuilders' harmonic torque data used in the torsional vibration calculations, *see* 2.2.3.

~~*2.1.3*~~ *2.1.4* Details of operating conditions encountered in service for prolonged periods, e.g. idling speed, trawling revolutions per minute, combinatory characteristics for installations equipped with controllable pitch propellers.

Existing paragraphs 2.1.4 *to* 2.1.6 *have been renumbered* 2.1.5 *to* 2.1.7*.*

2.2 Scope of calculations

2.2.3 The calculations carried out on oil engine systems are to be based on the Enginebuilders' harmonic torque data. (On request, Lloyd's Register (hereinafter referred to as 'LR') can provide a table of generalised harmonic torque components for use where appropriate.) The calculations are to take account of the effects of engine malfunction commonly experienced in service, such as a cylinder not firing (i.e. no injection but with compression), giving rise to the highest torsional vibration stresses in the shafting. Calculations are also to take account of a degree of imbalance between cylinders, characteristic of the normal operation of an engine under service conditions.

■ *Section 3* Design

3.1 Symbols and definitions

3.1.1 The symbols used in this Section are defined as follows:

d = minimum diameter of shaft considered, in mm

d_i = diameter of internal bore, in mm

k = the factor used in determining minimum shaft diameter, defined in Pt 11, Ch 2,4.2.1 and 4.4.3

r = ratio N/N_S or N_C/N_S whichever is applicable

C_d = a size factor defined as 0,35 + 0,93$d^{-0,2}$

C_k = a factor for different shaft design features, *see* Table 1.3.1

N = engine speed, in rev/min

N_c = critical speed, in rev/min

N_S = maximum continuous engine speed, in rev/min, or, in the case of constant speed generating sets, the full load speed, in rev/min

Q_S = rated full load mean torque

σ_u = specified minimum tensile strength of the shaft material, in N/mm^2

τ_c = maximum value of the vibration stress for continuous running at or below the maximum speed, in N/mm^2

τ_t = permissible stress due to torsional vibrations for transient operation, in N/mm^2

e = slot width, in mm

l = slot length, in mm.

~~σ_u = specified minimum tensile strength of the shaft material, in N/mm^2~~

~~C_k = a factor for different shaft design features, *see* Table 1.3.1~~

~~C_d = a size factor defined as 0,35 + 0,93$d^{-0,2}$~~

~~k = the factor used in determining minimum shaft diameter, defined in Pt 11, Ch 2,4.2.1 and 4.4.3.~~

~~**Table 1.3.1 C_k factors**~~

For intermediate shafts with			For thrust shafts external to engines		For propeller shafts
Integral coupling flanges	Shrink fit couplings	Keyways	On both sides of thrust collar	In way of axial bearing where a roller bearing is used as a thrust bearing	For which k = 1,22 and = 1,26
1,0	1,0	0,60	0,85	0,85	0,55
NOTE The determination of C_k - factors for shafts other than shown in this Table is at the discretion of LR.					

Table 1.3.1 C_k factors

Intermediate shafts with	
Integral coupling flange and straight sections	1,0
Shrink fit coupling	1,0
Keyway, tapered connection	0,60
Keyway, cylindrical connection	0,45
Radial hole	0,50
Longitudinal slot	0,30 (*see* 3.1.4)
Thrust shafts external to engines	
On both sides of thrust collar	0,85
In way of axial bearing where a roller bearing is used as a thrust bearing	0,85
Propeller shafts	
Flange mounted or keyless taper fitted propellers	0,55
Key fitted propellers	0,55
Between forward end of aft most bearing and forward stern tube seal	0,80
NOTE The determination of C_k factors for shafts other than shown in this Table will be specially considered by LR.	

3.1.4 For a longitudinal slot, C_k = 0,3 is applicable within the dimension limitations given in Pt 11, Ch 2,4.2.7. If the slot dimensions are outside these limitations, or if the use of another C_k is desired, the actual stress concentration factor (*scf*) is to be documented or determined from 3.1.5 or by direct application of FE calculation, in which case:

$$C_k = \frac{1{,}45}{scf}$$

Note that the *scf* is defined as the ratio between the maximum local principal stress and $\sqrt{3}$ times the nominal torsional stress (determined for the bored shaft without slots).

3.1.5 **Stress concentration factor of slots**. The stress concentration factor (*scf*) at the ends of slots can be determined by means of the following empirical formulae:

$$scf = \alpha_{t(\text{hole})} + 0{,}8\,\frac{\dfrac{(l-e)}{d}}{\sqrt{\left(1-\dfrac{d_i}{d}\right)\dfrac{e}{d}}}$$

This formula applies to:

- Slots at 120 or 180 or 360 degrees apart.
- Slots with semicircular ends. A multi-radii slot end can reduce the local stresses, but this is not included in this empirical formula.
- Slots with no edge rounding (except chamfering), as any edge rounding increases the *scf* slightly.

$\alpha_{t(hole)}$ represents the stress concentration of radial holes and can be determined as:

$$\alpha_{t(hole)} = 2{,}3 - 3\frac{e}{d} + 15\left(\frac{e}{d}\right)^2 + 10\left(\frac{e}{d}\right)^2\left(\frac{d_i}{d}\right)^2$$

where, in this context, e = hole diameter, in mm (this is independent of slot width)
or simplified to $\alpha_{t(hole)} = 2{,}3$.

3.2 Limiting stress in propulsion shafting

3.2.1 The following stress limits apply to intermediate shafts, thrust shafts and to screwshafts fully protected from seawater. For screwshafts, the limits apply to the ~~minimum section between the forward end of the propeller boss and the forward stern gland~~ portions of the screwshaft as defined in Pt 11, Ch 2,4.4.

3.5 Other machinery components

(*Part only shown*)
3.5.3 **Gearing:**
(a) The torsional vibration characteristics are to comply with the requirements of 2.2. The sum of the mean and of the vibratory torque should not exceed ~~one-third~~ four-thirds of the full transmission torque, at MCR, throughout the speed range. In cases where the proposed transmission torque loading on the gear teeth is less than the maximum allowable, special consideration will be given to the acceptance of additional vibratory loading on the gears.

3.6 Restricted speed and/or power ranges

3.6.1 Restricted speed and/or power ranges will be imposed to cover all speeds where the stresses exceed the limiting values, τ_c, for continuous running, including one cylinder misfiring conditions if intended to be continuously operated under such conditions. For controllable pitch propellers with the possibility of individual pitch and speed control, both full and zero pitch conditions have to be considered. Similar restrictions will be imposed, or other protective measures required to be taken, where vibratory torques or amplitudes are considered to be excessive for particular machinery items. At each end of the restricted speed range the engine is to be stable in operation.

3.6.2 The restricted speed range is to take account of the tachometer speed tolerances at the barred speeds.

Existing paragraph 3.6.2 *has been renumbered* 3.6.3*.*

~~*3.6.3*~~ *3.6.4* ~~Where shafting stresses due to a torsional critical response exceed the limiting values, τ_c, for continuous running, the speed restriction will be from:~~ Provided that the stress amplitudes due to a torsional critical response at the borders of the barred speed range are less than τ_c under normal and stable operating conditions, the speed restriction derived from the following formula may be applied:

$$= \frac{16}{18 - r} N_c \text{ to } \frac{18 - r}{16} N_c \text{ inclusive.}$$

Existing paragraphs 3.6.4 *to* 3.6.9 *have been renumbered* 3.6.5 *to* 3.6.10*.*

3.6.11 Restricted speed ranges in one-cylinder misfiring conditions on ships with single engine propulsion are to enable safe navigation whereby sufficient propulsion power is available to maintain control of the ship.

~~*3.6.10*~~ *3.6.12* There are to be no restricted speed ranges imposed above a speed ratio of $r \geq 0{,}8$ under normal operating conditions.

■ *Section 4*
Measurements

4.1 General requirements

4.1.3 The method of measurement is to be appropriate to the machinery components and the parameters which are of concern. Where shaft stresses have been estimated from angular amplitude measurements, and are found to be close to limits as defined in 3.2, strain gauge techniques may be required. When measurements are required, detailed proposals are to be submitted.

Cross-References

Section numbering in brackets reflects any Section renumbering necessitated by any of the Notices that update the current version of the Rules for Special Service Craft.

Part 11, Chapter 2

3.1.3 *now* 3.1.6	Reference to Part 11, Chapter 2, 3.1.2 *now reads* Part 11, Chapter 2, 3.1.3

Part 13, Chapter 1

3.6.8 *now* 3.6.9	Reference to Part 13, Chapter 1, 3.6.3 *now reads* Part 13, Chapter 1, 3.6.4

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